

EXPERIMENTAL VERIFICATION OF PARAMETERS IN AUTOMOBILE CRANKSHAFT MODELLING FOR VIBRATION ANALYSIS

Sanda Pyae Sone^{1,a}, Aung Lwin Moe^{2,b}, B.A. Aminudin^{1,a}, Oo Kyaw Nyein^{1,a}

Intelligence Dynamics System iKohza, MJIT, Universiti Teknologi Malaysia¹,

philip.sanda@gmail.com, aminudin.kl@utm.my, kokyawnyein@gmail.com

Department Mechanical Precision Engineering, MJIT, Universiti Teknologi Malaysia²,
aunglm@utm.my^b

ABSTRACT

In the interest of utilized more stable automobile components at high speed for reduction the vibration of mechanical system, dynamic characteristics analysis plays a vital role in complex mechanical parts. This paper introduces a clarified approach on statistical investigation and modal analysis methodology to study, predict and accurate crankshaft natural frequencies by using design of experiment (DOE). In this research, first, simulation had been done with MSC Nastran/ Patran to find out the natural frequencies in each mode shape of crankshaft as well as the verification with experiment was carried out. In order to less inaccuracy, numerous simplified crankshaft models were created by using these as input and DOE was established to acquire precise parameters of optimized crankshaft design as the second phase. This method can be further used for the optimizing the structural parameters and would provide some value basis to qualitative measure of parameters and determination of optimized structure. In Conclusion, modal verification accuracy between experimental and simulation has improved.

Keywords— Natural Frequency, Structure Modification, Design of Experiment, Crankshaft, Modal Data Analysis

1. INTRODUCTION

Recently, developments in design structure have always been an important issue and is performed with consecutive exposure of engineers. In the interest of advanced industries and manufacturing requirements, better solutions have been challenged to less in time consuming, resources and expanses. One of the important issues in automobile engineering, internal combustion engines could be seen on the vibrations and vibration forces and moments apply to engine components. These expectations motivate to observe preferred engine design and urge to optimize engine components especially in crankshaft production industry. During engine operation, the structure of crankshaft subjected to vibration can damage because of material fatigue resulting from the cyclic variation of the induced stress. Whenever the natural frequency of vibration of crankshaft coincided with the frequency of the external excitation from engine components, the engine system produce intense vibration known as resonance, which leads to excessive deflections and failure. Because of devastating

effects that vibrations can have on crankshaft, dynamic optimization has become a typical procedure in the design and development of most engineering systems.

In order to examine crankshaft structural design, various approaches and methods are carried out to reduce the vibration to keep apart from major source. A. Solanki^[1] reviewed that the comparative studies of design and optimization such as stress analysis, material and manufacturing processes, failure analysis, dynamic load analysis and design consideration as well as computer aided analysis in addition to cost reduction. Previously, Wang, Z.Q^[2] et. al., adopted finite element method to establish the dynamic model of crankshaft and provide technical support for the structural design. Mohammadi, M^[3] et. al., studied the modal analysis of crankshaft that is used in Samand Engine and showed that crankshaft has not a resonance phenomenon in the range of work experience. MC Cevik^[4] examine simplified methodology for selecting control factors to find out torsional stiffness and stress concentration factors (SCF) and that can show some predictability of the outputs of multibody analysis. Aminudin, ABU^[5] analyzed vibration level can be improved by selecting influence factors from design of experiment which enlightened algorithm of optimization for the crankshaft structure. Therefore, examination of crankshaft dynamic characteristic is essential in term of performance of the crankshaft itself at the early step of design.

Herein, the study aims to verify the parameters in experimental analysis in crankshaft modelling. At first, crankshaft was modelled in MSC Patran and studied natural frequencies based on major parameters. Experimental set up had been performed to obtain crankshaft frequencies and verified modal data. Eventually, the accuracy between these two analyses can be improved by using design of experiment of parameters.

2. DYNAMIC CHARACTERISTICS OF CRANKSHAFT

Crankshaft performance and stability under excitations from gas pressure is mainly depending on the vibration of crankshaft. Crankshaft vibration can be occurred due to torsional or bending deformation^[6]. In this research, we try to find out the dynamic behaviors of

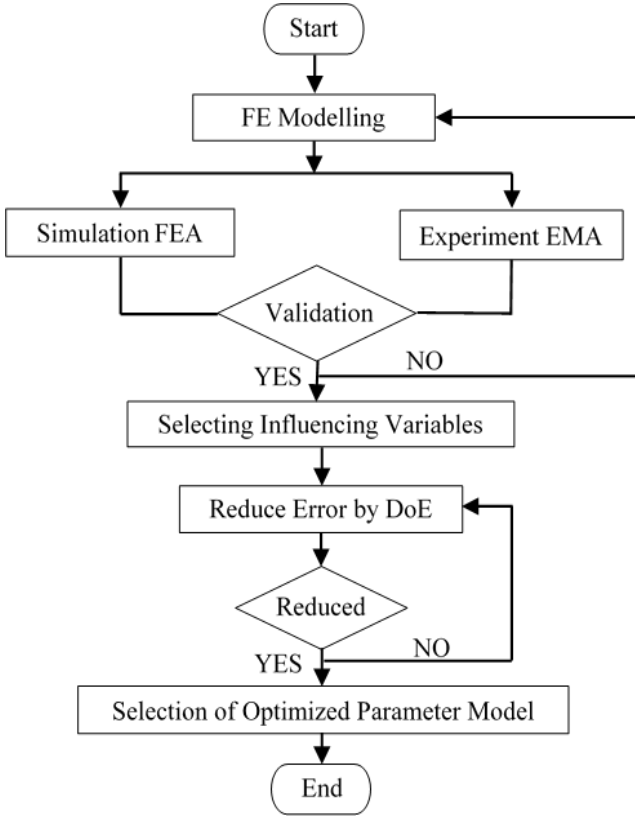


Figure 1. Flow chart of Study

crankshaft i.e. natural frequencies, mode shapes and frequency response functions. In this study, natural frequencies of crankshaft are focused. A methodology, as illustrated in Figure 1, is followed the procedure in order to stabilize a genuine design methodology and accuracy optimization process of crankshaft.

In Figure 1, the methodology improvement consists of, studying dynamic characteristics of crankshaft together with verification of experimental and simulation outcomes. The objectives of analysis can be summarized as; prediction of dynamic behavior of crankshaft, comparison of experimental and theoretical and reducing the error along with optimizing the simulation model. These investigation of above stated criteria will help to improve the accuracy of examination of multi body of freedom models and also accelerate the modal analysis of crankshaft by considering influencing factors.

2.1. Natural Frequencies

Most of objects while being hit or bumped or impacted or collide or somehow disturbed, will vibrate. When the objects vibrates, they will tend to vibrate at a particular frequency or a set of frequencies, which is called natural frequency of the object. Each degree of freedom of an object has its own natural frequency, expressed as ω_n . The speed of vibration divided by wavelength is known as frequency. Natural frequency can be either undamped or damped, depending on whether that system has significant damping.

Since the crankshaft has multi degree of freedom, the natural frequencies of such system is considered as beam model. The kinetic energy KE, for 'n' degree of freedom system can be expressed as

$$KE_e = \frac{1}{2} \int_V \dot{\mathbf{u}}^T \dot{\mathbf{u}} \rho dv \quad (1)$$

where, ρ = density (mass per unit volume), $\{\dot{\mathbf{u}}\}$ = velocity vector

In finite element, we divided the system into elements and each elements, were expressed $\{\dot{\mathbf{u}}\}$ in terms of the nodal displacement $\{q\}$, using shape function N. Thus, element kinetic energy is as follows,

$$KE_e = \frac{1}{2} \{\dot{q}\}_e^T m_e \{\dot{q}\}_e \quad (2)$$

The potential energy, PE of elastic system in finite element method can be expressed as

$$PE = \frac{1}{2} \{q\}_e^T k_e \{q\}_e - \{q\}_e^T \{f\}_e \quad (3)$$

The equation of motion for multiple-degrees-of-freedom systems can also be derived by using the Lagrangian Approach of analytical dynamics, which is viewed often as a preferred technique when complicating factors of geometry, kinematics, or modeling are present. Using the Lagrangian, $L = KE - PE$, the equation of motion is

$$M_e \{\ddot{q}\}_e + k_e \{q\}_e = \{f\}_e \quad (4)$$

Since, the mass, stiffness, force and displacement matrices give expression in local coordinate, after the superposition of all transformed finite element matrices mass, stiffness, force and displacement can be assemble and equation of motion can be rewritten as

$$[M]\{\ddot{q}\} + [K]\{q\} = \{F\} \quad (5)$$

where, $[M]$ = mass matrix (kg), $[K]$ = stiffness matrix (N/m), $\{F\}$ = force vector.

When the external force is equal to zero and considering a steady state, the solution becomes the eigenvalues problem. The characteristics of the equation is

$$\det | -\omega^2 [M] + [K] | = 0 \quad (6)$$

Thus, ω , natural frequencies of crankshaft can be determined.

3. CRANKSHAFT SYSTEM FINITE ELEMENT (FE) MODEL

Converting the crankshaft system into FE model for provide the precision of this analysis research. A layout of a crankshaft system is a complex structure and modelling of the crankshaft is being simplified as much as possible without affecting the reality of origin approach vibration analysis. Figure 2 demonstrates the modelling of crankshaft geometry. Beam crankshaft model is constructed in MSC Patran CAD software and clarified as cross-sectional circular and rectangular beams to find out the dynamic behavior of the structure. Unlike, 3D mesh model, beam model simplifies geometry and describe conventional form. That simulation model is composed of total 53 elements and total 54 nodes. Natural frequencies of crankshaft is computed as shown in Table 1. Total eight natural frequencies modes are found and commonly bending, torsion and twisting modes.

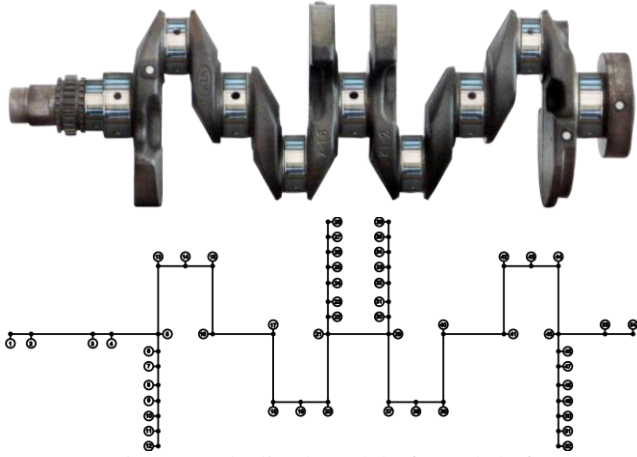


Figure 2. Idealized Model of Crankshaft

Table 1. Natural frequencies of Simulation Model

Mode	Natural Frequencies (Hz)	Description
1	400.96	Bending (Outplane)
2	412.34	Bending (Inplane)
3	774.64	Bending (Inplane)
4	802.62	Bending (Inplane)
5	985.81	Bending (Outplane)
6	1075.45	Bending (Outplane)
7	1170.98	Twisting
8	1548.94	Bending (Outplane)

4. EXPERIMENTAL MODAL ANALYSIS (EMA)

Experimental modal analysis (EMA) becomes well-known since the advent of the digital FFT spectrum analyzer in the early 1970's^[7]. This experimental method is commonly used to verify the result of analytical approach. However, sometimes, the modal parameters have to determine experimentally instead of analytical model. Since, impact testing is implied when it comes to find the modes of machines and structures because of its efficient method. Experiment was set up as shown in Figure 3. During this process, impact hammer, accelerometer, FFT analyzer and post-processing modal software were involved. The crankshaft was divided into 30 nodes and the experiment was set up by placing a crankshaft in free-free end condition with support of sponge. Triaxial accelerometer was put on crankshaft surfaces to detect responses while impact hammer is applied impulse to crankshaft. FRF data are collected using FFT analyzer and measurements from every impact have been taken and saved in LMS Software. From these measurements, natural frequencies form the EMA are as shown in Table 2. According to experimental outcomes, natural frequencies show good agreement between simulation modal and experimental model.

5. DESIGN OF EXPERIMENTS

Design of experiment is systematic and accurate approach to data collection to ensure the creation of robust

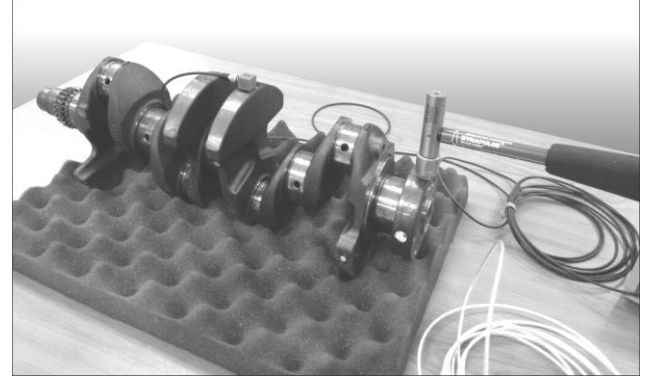


Figure 3. Experimental Setup

Table 2. Natural frequencies of Experimental Model

Mode	Natural Frequencies of Experiment (Hz)
1	355.48
2	456.73
3	779.12
4	804.91
5	919.83
6	1065.99
7	1242.02
8	1385.19

engineering conclusions. This method is utilized as solution in comparative, characterizing, modeling and optimizing areas. In this research, we focused in optimizing case; that is, determining optimal settings for each influencing factor and level that optimize the process response.

Based on a Taylor Series approximation, the deviation of the quality characteristic on either side of the target will cause to increase the quadratic loss function. In manufacturing process, product waste can be reduced by producing as similar as to the typical output. On the other hand, accuracy in a deviation to one side of the target may be more difficult than the other side. That's why, the simplicity and development of quadratic loss function appears in evaluating a deviation from the target as well as in ease of implementation. In order to measure the quality of all products, an improved evaluation approach should applicable in both within and outside specifications.

A quantitative evaluation of loss caused by functional variation can be derived as follows; $L_1(y)$ is differentiable function in the neighborhood of the target, y_0 . Using Taylor's series expansion, we have

$$L_1(y) = L_1(y_0) + L'_1(y_0)(y - y_0) + (L''_1 \frac{(y_0)(y-y_0)^2}{2!}) + \dots \quad (6)$$

Supposing the minimum quality loss at y_0 , and hence $L'_1(y_0) = 0$. Since $L_1(y_0)$ is a constant quality loss at y_0 , we defined the deviation loss of y from y_0 as

$$L_1(y) = L_1(y_0) = L'_1(y_0) = L''_1 \frac{(y_0)(y-y_0)^2}{2!} + \dots \quad (7)$$

Using a quadratic loss function, assume that these expansions approach to situations where y is close to , performance, y , deviates from the target, y_0 , and the loss associated with each product would be computed as follows;

$$L(y) = k(y - y_0)^2 \quad (8)$$

Where, k = quality loss coefficient

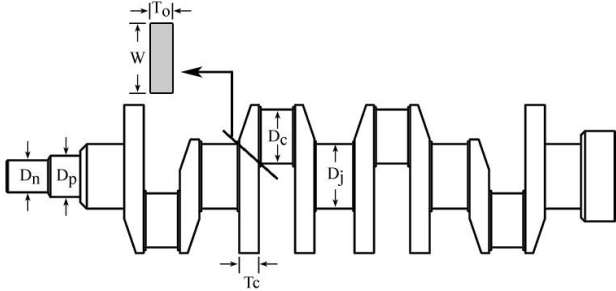


Figure 4. Selected Input Variables

In this research, in order to reduce the error between experimental and simulation model, “the smaller the better” quality characteristics was applied. Usually the smallest possible value for such characteristic is zero, and then $y_0 = 0$ becomes target value. Therefore, an approximation of $L(y)$ is

$$L(y) = ky^2, y \geq 0 \quad (9)$$

Natural frequencies of a crankshaft mainly depend on various design parameters. Figure 4 illustrates that influence factors normally on crankshaft. Thus, journal bearing diameter (D_j) and crankpin diameter (D_c) play an important roles in crankshaft vibration. As well as, the thickness of counterweight (T_c) control the balancing of crankshaft and then the crank nose diameter (D_n) and pulley diameter (D_p) are selected because these factors also lead to vary natural frequencies of crankshaft. The final and significant factors of crankshaft are overlap thickness (T_o) and width (W) of web which is effective in dynamic performance of crankshaft. Furthermore, Young’s modulus also basically effect on stiffness in material properties of crankshaft.

The main purpose of experimental design is to set up a statistical method which would support to determine which input variables show great effect on the output. In this study, Taguchi DOE approach is utilized for combination of all factor levels in an orthogonal array manner. The model verification have to maintain the accuracy by changing the Young’s moduli which has an influential effect on the stiffness and other important parameters. The condition of DOE is expressed in terms of control factors which represents variable of parameters and levels that determined by varying factors to different steps. For instance, 2 levels for material and 3 levels for rest of the parameters are selected. Since this paper is focused to reduce the error of model verification, the smaller the better signal-to-noise ratio (S/N) is applied.

Table 3. Control Factors and Levels

Parameters	Levels		
	1	2	3
Young’s Modulus, E (N/mm2)	2E11	2.21E11*	
Journal Bearing Diameter, D_j (mm)	47.89	47.96*	48.05
Crankpin Diameter, D_c (mm)	41.92	42.02*	42.81
Counterweight Thickness, T_c (mm)	17.78	18.45*	19.66
Overlap Thickness, T_o (mm)	17.3	18.45*	18.51
Overlap Width, W (mm)	77.46	77.85*	79.36
Crankshaft Nose Diameter, D_n (mm)	25.90	25.97*	26.05
Crankshaft Pulley Diameter, D_p (mm)	29.96	30.03*	30.32

* Original

Based on equation (9), S/N ratio equation can be expressed as

$$Q = -10 \log \left[\frac{1}{n} \sum_{i=1}^n y_i^2 \right] \quad (10)$$

Where Q = S/N ratio (dB), y_i = measurement value, and it was defined as error performance index for the validation in this research. That equation express that the calculated index is smaller by the time S/N ratio is increased. Instead of computing all the levels of control factor, Taguchi method simplifies the requirement of analysis. According to this approach, for those one 2 level and seven 3 levels control factors, L_{18} ($2^1 \times 3^7$) orthogonal arrays has been used and require only 18 numbers of runs to be tested. By inserting the control factors at Table 3, 18 variations are required to be able to analyze. Hence, the output of verification can be obtained by running sequence on each control factor.

In order to evaluate the accuracy of model analysis between experimental and simulation, error performance index can be defined as the addition of the square of natural frequency differences between experiment and simulation divided by experimental natural frequency.

Then, the equation for computing error performance index in correlation of experimental and simulation natural frequencies is defined as followed,

$$S = \left[\frac{f_{1exp} - f_{1sim}}{f_{1exp}} \right]^2 + \left[\frac{f_{2exp} - f_{2sim}}{f_{2exp}} \right]^2 + \left[\frac{f_{3exp} - f_{3sim}}{f_{3exp}} \right]^2 + \dots + \left[\frac{f_{nexp} - f_{nsim}}{f_{nexp}} \right]^2$$

$$S = \sum_{i=1}^n \left[\frac{f_{iexp} - f_{isim}}{f_{iexp}} \right]^2 \quad (11)$$

Where, S = error performance index, F_{nexp} = natural frequency from experiment at mode n and F_{nsim} = natural frequency from simulation at mode n .

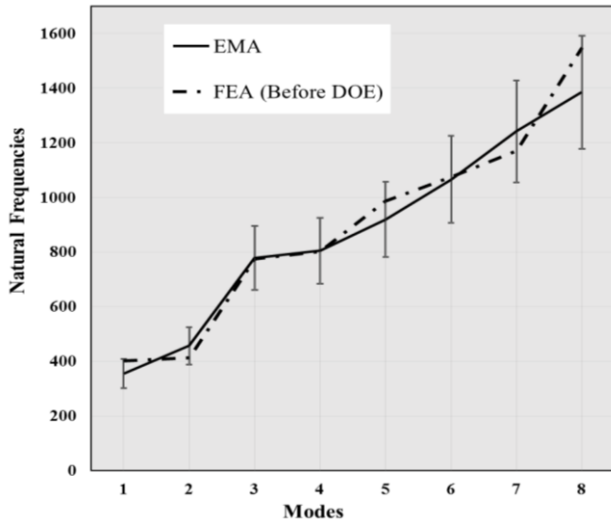


Figure 5. Natural Frequency Deviations of Experimental model from simulation model

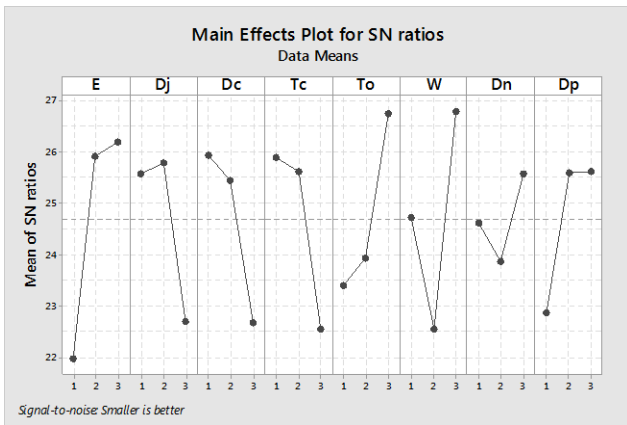


Figure 6. Main Factor Effects of S/N Ratio

6. RESULTS AND DISCUSSION

Correlation of natural frequencies is an important step to classify the identity of verification between simulation model and experimental model. As either FEA model has deficient meshing or EMA involves multiple dispositions, variations between models will always occur. This may increase to the probability of error in experimental data regarding with measurements and appearance of inherent model parameter error as well as model structure error.

Error deviations was performed and Figure 5 shows that the deviations of frequencies between FEA and EMA in each different modes of crankshaft. Symmetrical slope between these two models indicate that the mode frequencies were similar forms.

The encouraging approach based on orthogonal array experiments would simplify variances of control parameters along with optimum settings. The combination of design of experiment and optimization method express impressive outcomes as in Taguchi method. Signal-to-Noise ratio (S/N) is a log function of desire output and measures how the response varies with different noise conditions relative to target result. Depending on prediction of

Table 4. Optimum value of crankshaft

Control Factors	Before DOE	After DOE
Young Modulus	2.21E11	2.21E11
Journal Bearing Diameter	47.96	47.89
Crankpin Diameter	42.02	42.02
Counterweight Thickness	18.45	18.51
Overlap Thickness	18.45	18.45
Overlap Width,	77.85	77.85
Crankshaft Nose Diameter,	25.97	26.05
Crankshaft Pulley Diameter	30.03	30.03

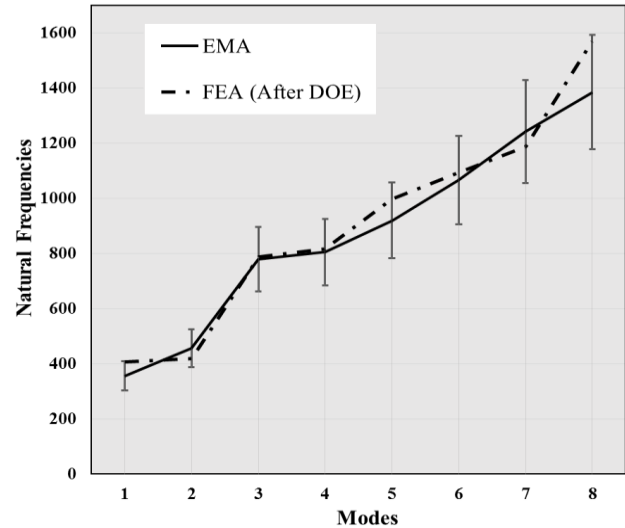


Figure 7. Error Deviation between EMA and FEA (after DOE)

optimum goal, this approach distribute as in objective data analysis.

As shown in Figure 6, the horizontal axis refers to the levels of influencing factors while the vertical axis shows S/N ratio level. The lower the effects of noise factors expect higher values of S/N ratio will appear. The control factors plot in Figure 6 shows in which levels are applicable for increasing S/N ratio. The optimum factor levels which should response with smallest error are the set point with the highest S/N ratio. According to the plot, the optimal parameters as peak level of S/N ratio which being selected are E – level 1 (Young's modulus), Dj – level 1 (journal bearing diameter, Dc – level 2 (crankpin diameter), Tc – level 2 (counterweight thickness), To – level 3 (overlap thickness), W – level 2 (overlap width), Dn – level 3 (crankshaft nose diameter) and Dp – level 2 (crankshaft pulley diameter).

Deviation of Natural frequencies in crankshaft compared to EMA and FEA model after DOE shows an accuracy improvement. In each frequency, error percentage in experimental model and outcome of simulation model after DOE appear in less than 15%.

As a result, journal bearing diameter, counterweight thickness and crankshaft nose diameter show altered parameter. Table 4 describes the values of control factors determined in condition of before DOE and after DOE values of crankshaft by simulation. With the optimum

value of after DOE parameters, the deviation between the natural frequencies of experiment and simulation models shows improvement in correlation. The error verification of experimental in crankshaft modelling and simulation of after DOE parameters is expressed in Figure 7. In Table 5, the total error optimized by using Taguchi DOE is expressed. Due to the precise dimensions select for determining levels, crankshaft structure is not effectively varied for economic manufacturing. The deviation between natural frequencies is significantly reduced in inplane modes while outplane modes show less effect on it. This can cause due to method of exciting in experiment. Even though the error reduction has to become in less amount, this method can identify and improve the accurate optimized parameter with 1.26%.

Table 5. Error Reduced before DOE and after DOE

	Before DOE (Hz)	After DOE (Hz)
Mode 1	400.96	406.3
Mode 2	412.34	417.97
Mode 3	774.64	786.45
Mode 4	802.62	815.48
Mode 5	985.81	996.47
Mode 6	1075.45	1093.42
Mode 7	1170.98	1186.98
Mode 8	1548.94	1568
Total Error Optimized		1.26%

7. CONCLUSION

Our study leads to the following conclusion;

- (1) The dynamic characteristic of crankshaft beam model was predicted and also comparison of analytical modal investigation of crankshaft structure were verified.
- (2) Thus analysis error that usually emerges in the verification procedure due to imperfection of the model information can be avoided meanwhile selecting to best parameters could be determined.
- (3) As a result, developed algorithm of optimizing various parameters together with the crankshaft model was proposed to improve the accuracy of experimental verification analysis. The diversity of natural frequencies between experimental and simulation model was also reduced by 1.26%.

ACKNOWLEDGEMENT

This work was mainly supported by the MJIIT ASEAN Student Incentive Award, Malaysia – Japan International Institute of Technology (MJIIT), Universiti Teknologi Malaysia, Kuala Lumpur.

REFERENCES

- [1] Solanki, Amit, Ketan Tamboli, and M. J. Zinjuwadia. "Crankshaft Design and Optimization-A Review." *National Conference on Recent Trends in Engineering & Technology*. 2011.
- [2] Wang, Zeng Qiang, and Hai Fei Qiu. "Calculation and Analysis of Dynamic Characteristic for Four-Cylinder Engine's Crankshaft." *Advanced Materials Research*. Vol. 940. 2014.
- [3] Mohammadi, M., Ranjbarkohan, M., Asadi, M.R. Modal analysis of crankshaft of samand engine (2012) *Journal of Applied Sciences Research*, 8 (7), pp. 3304-3307.
- [4] Cevik, Mehmet Cagri, Martin Rebbert, and Franz Maassen. *A New Approach for Prediction of Crankshaft Stiffness and Stress Concentration Factors*. No. 2010-01-0949. SAE Technical Paper, 2010.
- [5] Aminudin b. Abu, Oh, Jae-Eung, Lee, Jung-Youn, Optimization of Vibration Level for A Crankshaft on Various Parameters using Taguchi Method, *Twelfth International Congress on Sound and Vibration*, 11~14 July 2005, Lisbon, Portugal
- [6] Khare, Saharash. *Development of Innovative Simulation and Testing Methodologies for Optimizing Crankshaft Design*. No. 2010-32-0070. SAE Technical Paper, 2010.
- [7] Schwarz, B. J., & Richardson, M. H. (1999). Experimental modal analysis. *CSI Reliability Week, Orlando, FL*, 1–12.
- [8] Zetty, R. M. S., et al. "Investigation of Sensitivity Effect Based on Mass and Stiffness Modification in Automobile Crankshaft." *Applied Mechanics and Materials*. Vol. 752. 2015.
- [9] Hörnlund, Mats, and Alexander Papazoglu. "Analysis and measurements of vehicle door structural dynamic response." (2005).
- [10] Singh, Amarjeet, Vinod Kumar Mittal, and Surjit Angra. "Strength Analysis and Optimization Methods for Four Cylinder Engine Crankshaft Based on CATIA and ANSYS." *Applied Mechanics and Materials*. Vol. 592. 2014.
- [11] Rodrigues, Alex, et al. *Experimental and Numerical Modal Analysis of 6 Cylinders Diesel Crankshaft*. No. 2011-36-0358. SAE Technical Paper, 2011.
- [12] Schwarz, Brian J., and Mark H. Richardson. "Experimental modal analysis." *CSI Reliability week 35.1* (1999): 1-12.
- [13] Azoury, C., et al. "Experimental and Analytical Modal Analysis of a Crankshaft." *IOSR Journal of Engineering 2.4* (2012): 674-684.
- [14] Schmidt, Stephen R., and Robert G. Launsby. *Understanding industrial designed experiments*. Air Academy Press, 1989.